Catalogue of

High Pressure Hydraulic
Fittings and Flanges

. . Folly's Patent . .

and

Information for Use in Designing

Hydraulic Plants

"Tight Joint" Elbow

Tight Joint Company

159-161 Bank Street
New York

CATALOGUE OF

HIGH PRESSURE HYDRAULIC

FITTINGS AND FLANGES.

(FOLLY'S PATENT.)

AND

INFORMATION FOR USE IN DESIGNING HYDRAULIC PLANTS.

BY

JOHN PLATT,

MEMBER OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS,
AND ASSOC. M. INST. C. E.

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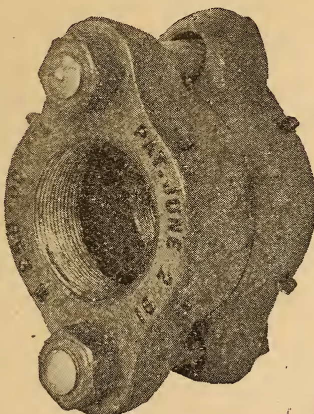


Leak.

"TIGHT JOINT" FLANGE.

TIGHT JOINT CO.
159-161 BANK STREET,
New York.

PREFACE.

In the following pages it has been our aim to place before the hydraulic engineering world particulars of our well-known "Tight Joint" and its adaptation to fittings and flanges specially designed to suit their needs.

We are glad to be able to report that engineers can now procure fittings and flanges that will insure their being able to make tight joints, and that the trouble and annoyance of leaky joints they have heretofore had to contend with, can be obviated by the use of our specialties.

For the compiling of the "Information for use in designing Hydraulic Plants" we are indebted to Mr. John Platt, the well-known high pressure hydraulic engineer, of 97 Cedar St., New York City. Much of the information is taken from notes collected by him, both in this country and while associated with Mr. James Platt, Member of Council of the Institute of Mechanical Engineers, who has done so much to further the development of high pressure hydraulic machinery. A number of the tables have been specially compiled and others taken from Mr. William Kent's "Mechanical Engineer's Pocket Book," Mr. Henry Adams's "Hand-book for Mechanical Engineers." and other engineering note-books.

We trust that the information given will be found of benefit to those who undertake the designing of hydraulic machinery, and we shall be glad if its users will kindly draw our attention to any errors of omission or commission, and also make any suggestions that may tend to make the "Information" more valuable.

ERRATA.

Page 27.

Table of W. I. & S. pipe should read,

No. of threads per inch.

18
18
14
14
11½
11½
11½
11½
8

Page 37.

Capacities of Cylinders,

Diam. Inches.	Load at 1500 lbs.
13	199050

Page 34.

Thickness of cylinder should read,

Rankine gives.

$$R = \sqrt{\frac{s + p}{s - p}} \times r$$

Page 39.

Seventh line from bottom of page should read.

Efficiency per cent.

80 76 72

Page 41.

Thirteenth line from top of page should read.

½″ pipe reduced to ⅜″ at one point.

Formulae for areas of valves should read,

W = Weight of ram, &c.

Page 43.

Formula should read.

$$a = \frac{w\ k^2\ f\ m}{l\ d\ p}$$

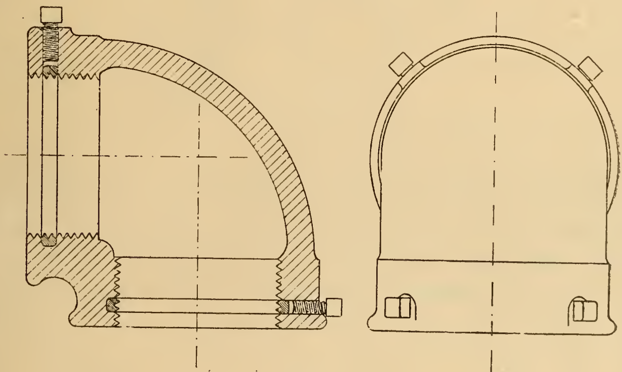
PART I.

HIGH PRESSURE HYDRAULIC FITTINGS AND FLANGES.

(FOLLY'S PATENT.)

Description of Tight Joint Hydraulic Fittings and Flanges.

"Tight Joint" fittings have established such a reputation for ammonia work, and proved so very satisfactory under the heaviest pressures obtainable, that we have decided to manufacture a line of Standard High Pressure Hydraulic Fittings and Flanges, making use of the best hydraulic experience of this country and England.

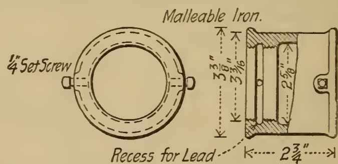


(1) SECTIONAL CUT OF FITTING.

The "Tight Joint" itself will be readily understood from Figure 1, and the following description:

The pipe is screwed into the fitting as in the case of ordinary low pressure work, and does not butt against a shoulder. The joint is made by means of a lead collar 1–4 in. wide by 3–16 in. deep, cast in the fitting three or four threads from the end.

Holes leading into the recess are tapped for 1–4 in. set screws, one or more set screws being used, according to the size of the fittings. The lead collar is formed on a mandrel of slightly smaller size than the fitting, so that the lead is left projecting a very little beyond the threads. If a pipe be screwed into the fitting it will expand the lead packing, causing it to tightly fill the screw threads, and if the joint is not made tight by this means, one or more turns of the compressing screw will crowd the lead around the pipe and make it absolutely and permanently tight.



(2) "TIGHT JOINT" COUPLING.

Having described our system of making screwed joints tight, we will now look into the question of high pressure fittings and compare the old practice of making joints with our own.

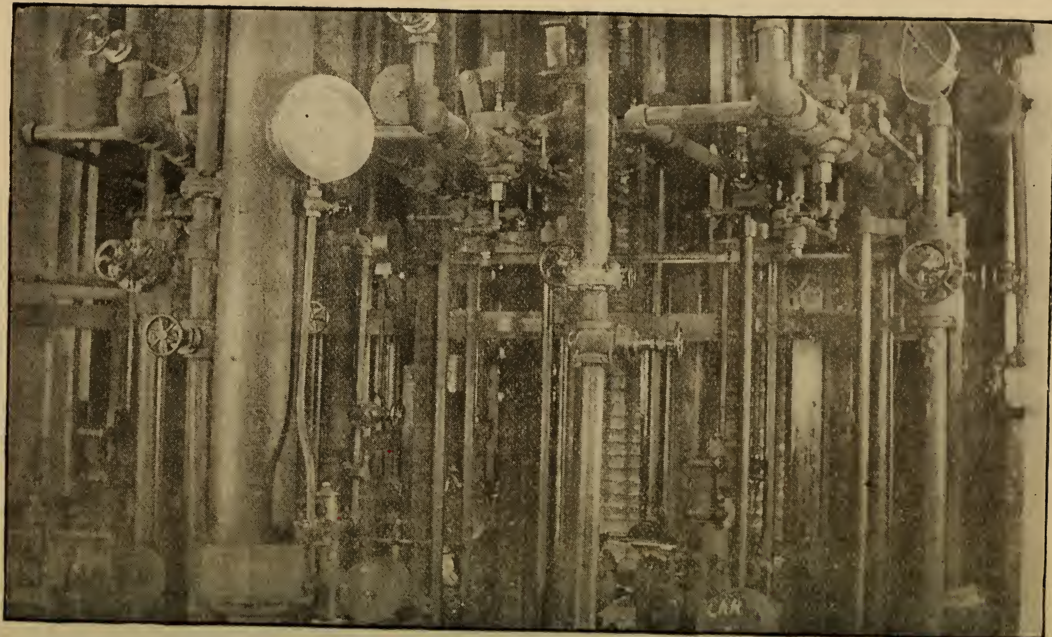
Two methods are adopted in general practice, the one depends upon screwing the pipe into the fitting so tightly that a leak cannot take place, this necessitates a perfect thread and also that very considerable force be used in screwing the pipe into the fitting, so much being necessary at times that it is no unusual thing to burst a fitting. The second method —a preferable one—is to let the end of the pipe butt against some material, such as leather or copper, and thus make a joint between the two pipes; in this case it is not necessary that the pipes be screwed so tightly into the fitting, but to insure a perfect joint under high pressure great care has to be taken in preparing the end of the pipe. In piping up a plant having a number of turns it is necessary that the pipes

shall be cut off absolutely to length; and in the case of flanges, in erection it is even necessary to take down one piece of pipe a number of times to alter the length so that the flange can be screwed around to bring the boltholes in their proper position.

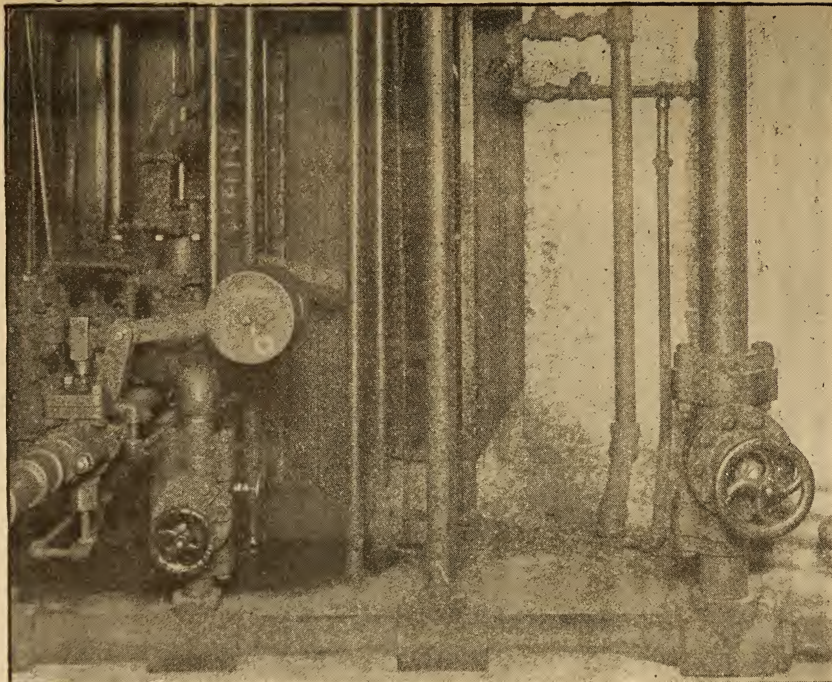
Those who have had to do with piping up for high pressure will appreciate the difficulties they encounter in erection and will at once see the advantage they will secure from using our type of fittings.

It will be seen from Figure 1 that in the case of the Tight Joint, as long as the edge of the pipe projects a quarter of an inch beyond the lead collar a perfect joint can be assured, but the end of the pipe may be carried as much as $\frac{3}{4}$ of an inch beyond this point. In this way a considerable latitude is possible and erection made very easy.

As an example of what has been done in this direction with Tight Joints, we would refer to the piping for the High Pressure Hydraulic Elevator Systems in the Girard Life Building in Philadelphia and the American Tract Society Building in New York, put in by Otis Brothers Company. In the latter plant several hundred fittings were used, and some idea of the intricacy of the piping may be obtained from Figures 3 and 4.



(3) Photograph showing nest of piping in the Otis High Duty Elevator Plant in American Tract Society Building.



(4) Photograph showing 5 inch pipe line under 750 lbs. working pressure in American Tract Society Building.

On testing these two plants (which carry a working pressure of 750 lbs.) not a single leaky joint was found.

A great many experiments have been made with the Tight Joint fittings and everything has been done that was possible to test their efficiency.

Prof. D. S. Jacobus, of the Stevens Institute, made a series of tests, and the following are some extracts from his report:

"½-inch and ¾ inch Brass Tees.—Made to leak and tighten up at 300 pounds per square inch steam pressure, 1,700 pounds per square inch gas pressure, and 3,000 pounds per square inch heavy refined petroleum oil pressure.

"1-inch Iron Tee, No. 2.—Made to leak and tighten up at 5,000 and 10,000 pounds crude petroleum oil pressure and 5,000 and 10,000 pounds water pressure. Withstood 15,000 pounds per square inch crude oil pressure without leakage and 16,000 pounds water pressure.

"1-inch Brass Hydraulic Tee, No. 2.—Made to leak and tighten up at 5,000 pounds per square inch water pressure."

From these experiments it will be seen that it is possible by tightening up the small set-screw to compress the lead and to make it flow in such a manner that a leak at five or six thousand pounds per square inch can be taken up and a joint made perfectly tight without taking the pressure off the system.

To show that expansion and contraction have no effect whatever upon the joints we would refer to the following report made by Mr. Charles O'Connor, superintendent of the Pratt Works of the Standard Oil Company:

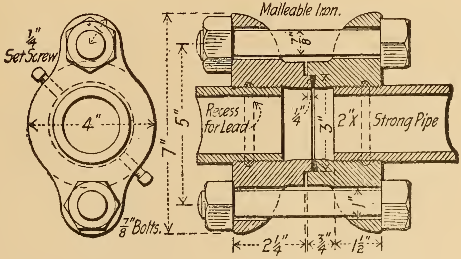
"I have tested your fitting and find it to be in every sense just as you recommend it. The tests were made in the following manner: I built a coil of 2-inch pipe, using six (6) of your 2-inch Return Bends, making 12 joints. On first test we found that at 300 pounds air pressure (under water) there were two small leaks, and by setting up one or two turns on set-screws they became tight. I was somewhat doubtful as to the joints remaining tight where there would be great expansion and contraction, so I raised coil out of water and turned steam through it at 80 pounds pressure until coil was

as hot as steam would make it, then I turned off steam, submerged coil in cold water and again applied 300 pounds air pressure and found every joint tight."

That the working of a large hydraulic system during an extended period has no effect upon the joint has been proved by its use in High Pressure Hydraulic Elevators, Hydraulic Riveters, Presses, General Hydraulic Systems, High Pressure Steam, 1,500 per □" Air Pressure and a very extended use in all classes of Ammonia Work.

All fittings and flanges are made of air furnace malleable iron having a tensile strength of from fifty to sixty thousand pounds per square inch. We do not use the so-called malleable iron poured from a cupola, and can thus insure fittings being sound and reliable.

All fittings are tested by hydraulic pressure before leaving the works, as per table on page 33.



(5) "TIGHT JOINT" FLANGE UNION;

The flange unions are furnished complete with bolts and guttapercha rings. Single flanges, either male or female, can be used for connecting to valves or cylinders. The tables of flanges on pages 15 and 16 will furnish the dimensions necessary for engineers to design the details of their connections to suit the fittings we carry in stock.

The joint between the flanges is the "Armstrong" standard and made, preferably, with a round guttapercha ring. This makes the best and most durable joint, though leather or lead can also be used for this purpose.

The threaded openings of the flange unions are made tight by the use of the "Tight Joint" lead collar and set-screws described on page 7. The combination of this device with the "Armstrong" flange joint makes an absolutely reliable hydraulic flange union.

The following tables give the standard sizes of Flanges for 750 and 1,500 lbs. per square inch.

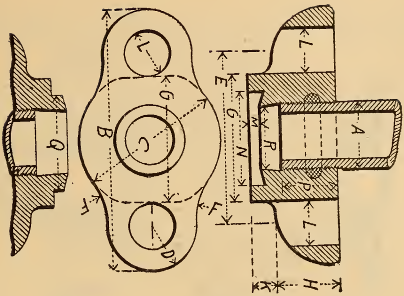


Table of "Tight Joint" Flanges.
750 lbs. Working Pressure.

Nominal Inside Diameter	A	B	C	D	E	F	G	H	K	L	M	N	O	P	Q	R	Size of Bolts	Number of Set-Screws
1/2	.84	3 7/8	2 1/4	11/16	2 1/2	1 1/4	1 3/4	1	3/8	11/16	3/16	1 1/4	1/4	7/8	1 7/32	3/4	5/8	1
3/4	1.05	4 1/8	2 1/4	11/16	2 3/4	1 1/4	2	1	3/8	3/4	3/16	1 1/2	1/4	7/8	1 15/32	1	5/8	1
1	1.31	4 1/2	2 1/2	3/4	3	1 1/4	2 1/4	1 1/8	3/8	3/4	1/4	1 3/4	1/4	1	1 23/32	1 3/16	5/8	1
1 1/4	1.66	5 1/8	2 7/8	13/16	3 1/2	1 3/8	2 5/8	1 1/4	1/2	7/8	1/4	2 1/8	5/16	1 1/8	2 3/32	1 1/2	3/4	2
1 1/2	1.9	5 3/8	3 1/4	13/16	3 3/4	1 3/4	2 7/8	1 3/8	1/2	7/8	1/4	2 3/8	5/16	1 1/4	2 11/32	1 3/4	3/4	2
2	2.37	7	4	1	5	2	3 7/8	1 1/2	3/4	1	1/4	3	5/16	1 1/2	2 31/32	2 1/4	7/8	2
2 1/2	2.87	7 5/8	4 3/4	1 1/8	5 3/8	2 1/4	4 1/8	1 5/8	3/4	1 1/8	5/16	3 1/4	3/8	1 1/2	3 7/32	2 1/2	1	2
3	3.5	8 1/2	5 1/4	1 1/8	6 1/4	2 1/2	5	1 3/4	3/4	1 1/8	5/16	4	3/8	1 1/2	3 31/32	3 1/4	1	4
4	4.5	9 7/8	6 1/2	1 5/16	7 1/4	3	5 7/8	2	3/4	1 3/8	5/16	4 7/8	7/16	1 3/4	4 27/32	4 1/8	1 1/4	5
5	5.5	11 5/8	7 3/4	1 9/16	8 1/2	3 1/2	7	2 1/4	3/4	1 1/2	3/8	6	1/2	2	5 31/32	5 1/4	1 3/8	6
6	6.62	13 1/4	9	1 5/8	10	4 1/2	8	2 1/2	1	1 5/8	3/8	7	1/2	2 1/4	6 31/32	6 1/4	1 1/2	7

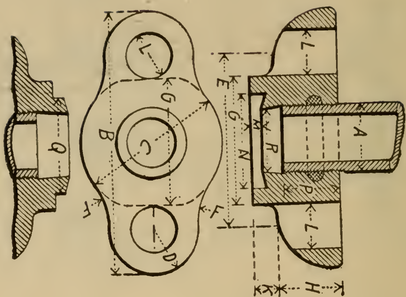


Table of Tight Joint Flanges.—1,500 lbs. Working Pressure.

Nominal Inside Diameter	A	B	C	D	E	F	G	H	K	L	M	N	O	P	Q	R	Size of Bolts	Number of Set-Screws
$\frac{3}{8}$	.67	$3\frac{1}{4}$	$1\frac{3}{4}$	$\frac{9}{16}$	$2\frac{1}{8}$	1	$1\frac{1}{2}$	$\frac{7}{8}$	$\frac{5}{16}$	$\frac{9}{16}$	$\frac{3}{16}$	1	$\frac{1}{4}$	$\frac{13}{16}$	$\frac{31}{32}$	$\frac{1}{2}$	$\frac{1}{2}$	1
$\frac{1}{2}$	.84	$3\frac{7}{8}$	$2\frac{1}{4}$	$\frac{11}{16}$	$2\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{3}{4}$	1	$\frac{3}{8}$	$\frac{11}{16}$	$\frac{3}{16}$	$1\frac{1}{4}$	$\frac{1}{4}$	$\frac{7}{8}$	$1\frac{7}{32}$	$\frac{3}{4}$	$\frac{5}{8}$	1
$\frac{3}{4}$	1.05	$4\frac{1}{4}$	$2\frac{1}{2}$	$\frac{11}{16}$	$2\frac{7}{8}$	$1\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{3}{16}$	$\frac{7}{16}$	$\frac{3}{4}$	$\frac{1}{4}$	$1\frac{1}{2}$	$\frac{1}{4}$	1	$1\frac{15}{32}$	1	$\frac{5}{8}$	1
1	1.31	$4\frac{7}{8}$	$2\frac{3}{4}$	$\frac{13}{16}$	$3\frac{1}{4}$	$1\frac{3}{4}$	$2\frac{3}{8}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{7}{8}$	$\frac{1}{4}$	$1\frac{3}{4}$	$\frac{5}{16}$	$1\frac{1}{8}$	$1\frac{23}{32}$	$1\frac{3}{16}$	$\frac{3}{4}$	1
$1\frac{1}{4}$	1.66	$5\frac{1}{4}$	$3\frac{1}{4}$	$\frac{13}{16}$	$3\frac{5}{8}$	2	$2\frac{3}{4}$	$1\frac{3}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$\frac{1}{4}$	$2\frac{1}{8}$	$\frac{5}{16}$	$1\frac{1}{8}$	$2\frac{3}{32}$	$1\frac{1}{2}$	$\frac{3}{4}$	2
$1\frac{1}{2}$	1.9	6	$3\frac{1}{2}$	$\frac{15}{16}$	$4\frac{1}{8}$	$2\frac{1}{4}$	$3\frac{1}{8}$	$1\frac{1}{2}$	$\frac{1}{2}$	1	$\frac{1}{4}$	$2\frac{3}{8}$	$\frac{5}{16}$	$1\frac{1}{4}$	$2\frac{11}{32}$	$1\frac{3}{4}$	$\frac{7}{8}$	2
2	2.37	7	$4\frac{1}{4}$	$1\frac{1}{16}$	$4\frac{7}{8}$	$2\frac{1}{4}$	$3\frac{3}{4}$	$1\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{1}{4}$	3	$\frac{3}{8}$	$1\frac{1}{2}$	$2\frac{31}{32}$	$2\frac{1}{4}$	1	2
$2\frac{1}{2}$	2.87	$7\frac{3}{4}$	$4\frac{3}{4}$	$1\frac{3}{16}$	$5\frac{3}{8}$	$2\frac{1}{2}$	$4\frac{1}{8}$	2	1	$1\frac{1}{4}$	$\frac{5}{16}$	$3\frac{1}{4}$	$\frac{3}{8}$	$1\frac{1}{2}$	$3\frac{7}{32}$	$2\frac{1}{2}$	$1\frac{1}{8}$	2

Sizes Carried in Stock for 750 lbs. Working Pressure.

COUPLINGS.	ELBOWS.	TEES.	FLANGE UNIONS.	REDUCING BUSHINGS.
1/4	1/4	1/4	—	1/4
3/8	3/8	3/8	—	3/8
1/2	1/2	1/2	1/2	1/2
3/4	3/4	3/4	3/4	3/4
1	1	1	1	1
1 1/4	1 1/4	1 1/4	1 1/4	1 1/4
1 1/2	1 1/2	1 1/2	1 1/2	1 1/2
2	2	2	2	2
2 1/2	2 1/2	2 1/2	2 1/2	2 1/2
3	3	3	3	3
4	4	4	4	4
5	5	5	5	5
6	6	6	6	6

Sizes Carried in Stock for 1,500 lbs. Working Pressure.

COUPLINGS.	ELBOWS.	TEES.	FLANGE UNIONS.	REDUCING BUSHINGS.
1/4	1/4	1/4	—	1/4
3/8	3/8	3/8	3/8	3/8
1/2	1/2	1/2	1/2	1/2
3/4	3/4	3/4	3/4	3/4
1	1	1	1	1
1 1/4	1 1/4	1 1/4	1 1/4	1 1/4
1 1/2	1 1/2	1 1/2	1 1/2	1 1/2
2	2	2	2	2
2 1/2	2 1/2	2 1/2	2 1/2	2 1/2

Sizes Carried in Stock for 3,000 lbs. Working Pressure.

COUPLINGS.	ELBOWS.	TEES.	FLANGE UNIONS.	REDUCING BUSHINGS.
$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$
$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
1	1	1	1	1
$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$
$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$
2	2	2	—	2
$2\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	—	$2\frac{1}{2}$
4	4	4	—	4
	6	6	—	6

Sizes Carried in Stock for 5,000 lbs. Working Pressure.

COUPLINGS.	ELBOWS.	TEES.	FLANGE UNIONS.	REDUCING BUSHINGS.
$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$
$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
1	1	1	1	1
$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$
$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$

The Reducing Bushings have the lead collar and set-screws the same as our regular fittings and can be supplied to meet any desired changes in size.

The Flange Unions are in pairs (male and female) furnished complete with bolts and pure guttapercha rings.

We can make to order Flanges and Fittings to stand any desired pressure.

PART II.

INFORMATION

FOR USE

IN DESIGNING HYDRAULIC PLANTS.

COMPILED BY

JOHN PLATT, Member of the American Society of Mechanical Engineers,
Assoc. M. Inst. C. E.

INTRODUCTION.

The following tables, formulæ and general information have been compiled in the hope that they will be found of use by those who use hydraulic fittings and have to deal with the general question of hydraulic pressure transmission.

The tables of wrought iron and steel pipe are for standard sizes. We have given at the head of each table pressures for which the various thicknesses of pipe can be used. The makers, as a general rule, will not guarantee their pipe for any pressure, but pipe of a good quality from a reliable maker will be found to be perfectly good for the pressures stated.

With regard to formula for thickness of hydraulic cylinders, considerable discretion is necessary when using any formula for thick cylinders, and unless those attempting to use such formula have considerable shop experience, we would recommend the taking of the thickness from the table used by Sir W. G. Armstrong.

Circumferences, Areas, Squares, Etc.,

ADVANCING BY DECIMALS—.1 to 9.8.

Diameter.	Circumference.	Area.	Square.	Cube.	Square Root.	Cube Root.
.1	.314	.00785	.01	.001	.316	.464
.2	.628	.0314	.04	.008	.447	.585
.3	.942	.0706	.09	.027	.548	.669
.4	1.26	.1256	.16	.064	.633	.737
.5	1.57	.1963	.25	.125	.707	.794
.6	1.88	.2827	.36	.216	.775	.843
.7	2.20	.3848	.49	.343	.837	.888
.8	2.51	.5026	.64	.512	.894	.928
.9	2.83	.6362	.81	.729	.949	.965
1.	3.14	.7854	1.	1.	1.	1.
.1	3.46	.9503	1.21	1.33	1.049	1.032
.2	3.77	1.131	1.44	1.73	1.095	1.063
.3	4.08	1.327	1.69	2.20	1.140	1.091
.4	4.39	1.539	1.96	2.74	1.183	1.119
.5	4.71	1.767	2.25	3.37	1.225	1.145
.6	5.02	2.011	2.56	4.10	1.265	1.170
.7	5.34	2.270	2.89	4.91	1.304	1.193
.8	5.65	2.545	3.24	5.83	1.342	1.216
.9	5.96	2.835	3.61	6.86	1.378	1.239
2.	6.28	3.142	4.	8.	1.414	1.260
.1	6.59	3.464	4.41	9.26	1.449	1.281
.2	6.91	3.801	4.84	10.65	1.483	1.301
.3	7.22	4.155	5.29	12.17	1.517	1.320
.4	7.53	4.524	5.76	13.82	1.549	1.339
.5	7.85	4.909	6.25	15.63	1.581	1.357
.6	8.16	5.309	6.76	17.58	1.612	1.375
.7	8.48	5.726	7.29	19.68	1.643	1.392
.8	8.79	6.158	7.84	21.95	1.673	1.409
.9	9.11	6.605	8.41	24.39	1.703	1.426
3.	9.42	7.069	9.	27.	1.732	1.442
.2	10.05	7.548	10.24	32.77	1.789	1.474
.4	10.68	8.553	11.56	39.30	1.844	1.504

Circumferences, Areas, Squares, Etc.,

ADVANCING BY DECIMALS. .1 to 9.8.—*Continued.*

Diameter.	Circumference.	Area.	Square.	Cube.	Square Root.	Cube Root.
3.6	11.30	10.18	12.96	46.66	1.897	1.533
.8	11.93	11.34	14.44	54.87	1.949	1.560
4.	12.56	12.57	16.	64.	2.	1.587
.2	13.19	13.85	17.64	74.09	2.049	1.613
.4	13.82	15.21	19.36	85.18	2.098	1.639
.6	14.45	16.62	21.16	97.34	2.145	1.663
.8	15.08	18.10	23.04	110.6	2.191	1.687
5.	15.70	19.63	25.	125.	2.236	1.710
.2	16.33	21.24	27.04	140.6	2.280	1.732
.4	16.96	22.90	29.16	157.5	2.324	1.754
.6	17.59	24.63	31.36	175.6	2.366	1.776
.8	18.22	26.42	33.64	195.1	2.408	1.797
6.	18.84	28.27	36.	216.	2.449	1.817
.2	19.47	30.19	38.44	238.3	2.490	1.837
.4	20.10	32.17	40.96	262.1	2.530	1.857
.6	20.73	34.21	43.56	287.5	2.569	1.876
.8	21.36	36.32	46.24	314.4	2.608	1.895
7.	21.99	38.48	49.	343.	2.646	1.913
.2	22.61	40.72	51.84	373.2	2.683	1.931
.4	23.24	43.01	54.76	405.2	2.720	1.949
.6	23.87	45.36	57.76	439.	2.757	1.966
.8	24.50	47.78	60.84	474.6	2.793	1.983
8.	25.13	50.27	64.	512.	2.828	2.
.2	25.76	52.81	67.24	551.4	2.864	2.017
.4	26.38	55.42	70.56	592.7	2.898	2.033
.6	27.01	58.09	73.96	636.1	2.933	2.049
.8	27.64	60.82	77.44	681.5	2.966	2.065
9.	28.27	63.62	81.	729.	3.	2.080
.2	28.90	66.48	84.64	778.7	3.038	2.095
.4	29.53	69.40	88.36	830.6	3.066	2.110
.6	30.15	72.38	92.16	884.7	3.098	2.125
.8	30.78	75.43	96.04	941.2	3.130	2.140

Wrought-Iron and Steel Pipe.

For Working Pressures up to 300 lbs. per Square Inch.

TABLE OF STANDARD SIZES AND DIMENSIONS.

Nominal Inside Diam.	Actual Inside Diam.	Actual Outside Diam.	Thick-ness.	Internal Area.	Nominal weight per foot of length.	No. of Threads per inch of screw.	Contents of one foot in length.
Inches.	Inches.	Inches.	Inches.	Sq. Inches.	Pounds.		Gallons.
$\frac{1}{4}$	.36	.54	.08	.10	.42	18	.005
$\frac{3}{8}$	.49	.67	.09	.19	.56	18	.009
$\frac{1}{2}$	.62	.84	.10	.30	.84	14	.015
$\frac{3}{4}$	.82	1.05	.11	.53	1.12	14	.027
1	1.04	1.31	.13	.86	1.67	14	.044
$1\frac{1}{4}$	1.38	1.66	.14	1.49	2.24	$11\frac{1}{2}$	.075
$1\frac{1}{2}$	1.61	1.9	.14	2.03	2.68	$11\frac{1}{2}$	.105
2	2.06	2.37	.15	3.35	3.61	$11\frac{1}{2}$	.173
$2\frac{1}{2}$	2.46	2.87	.20	4.78	5.74	$11\frac{1}{2}$	.248
3	3.06	3.5	.21	7.38	7.54	8	.383
4	4.02	4.5	.23	12.73	10.66	8	.661
5	5.04	5.56	.25	19.99	14.50	8	1.03
6	6.06	6.625	.28	28.88	18.76	8	1.50

Extra Strong—Wrought-Iron and Steel Pipe.

WROUGHT IRON FOR WORKING PRESSURE UP TO 1,000 LBS. PER SQUARE INCH.
STEEL OF GOOD QUALITY " " " " " 1,500 " " " "

TABLE OF STANDARD SIZES AND DIMENSIONS

Nominal Inside Diam.	Actual Inside Diam.	Actual Outside Diam.	Thickness.	Internal Area.	Nominal weight per foot of length.	Contents of one foot in length.
Inches.	Inches.	Inches.	Inches.	Sq. inches.	Pounds.	Gallons.
1/4	.29	54	.12	.06	.54	.003
3/8	.42	.67	.12	.13	.74	.006
1/2	.54	.84	.14	.23	1.09	.011
3/4	.73	1.05	.15	.45	1.53	.023
1	.95	1.31	.18	.71	2.17	.036
1 1/4	1.27	1.66	.19	1.27	3.00	.065
1 1/2	1.49	1.9	.20	1.75	3 63	.090
2	1.93	2.37	.22	2.93	5.02	.152
2 1/2	2.31	2.87	.28	4.20	7.67	.218
3	2.89	3.50	.30	6.56	10.25	.340
4	3.81	4.50	.34	11.44	14.97	.594
5	4.81	5.56	.37	18.19	20.54	.944
6	5.75	6.625	.43	25.93	28.58	1.34

Double Extra Strong—Wrought-Iron and Steel Pipe.

For Working Pressures up to 5,000 lbs. per Square Inch.

TABLE OF STANDARD SIZES AND DIMENSIONS.

Nominal Inside Diam.	Actual Inside Diam.	Actual Outside Diam.	Thickness.	Internal Area.	Nominal weight per foot of length.	Contents of one foot in length.
Inches.	Inches.	Inches.	Inches.	Sq. Inches.	Pounds.	Gallons.
⅜	.23	.67	.22	.04	.96	.002
½	.24	.84	.29	.04	1.50	.002
¾	.42	1.05	.31	.13	2.30	.006
1	.58	1.31	.36	.27	3.40	.014
1¼	.88	1.66	.38	.61	5.00	.031
1½	1.08	1.9	.40	.93	6.45	.048
2	1.49	2.37	.44	1.74	9.48	.090
2½	1.75	2.87	.56	2.41	13.30	.125
3	2.28	3.50	.60	4.09	17.70	.212
4	3.13	4.50	.68	7.72	24.70	.401
5	4.06	5.56	.75	12.96	37.10	.629
6	5.06	6.625	.78	20.10	50.10	1.04

Table of Tensile Strength in Pounds, Per Square Inch.

Metals and Alloys.

Brass, Cast	Average	18,000
Bronze, or Gun Metal	36,000 to	50,000
Copper, Cast	Average	19,000
" Bolts	"	36,000
Iron, Cast, 13,500 to 29,000	"	16,500
" Malleable (Air furnace)	35,000 to	55,000
" Wrought	46,000 to	56,000
Steel, Cast	65,000 to	85,000
" Crucible	90,000 to	120,000
" Rivet	55,000 to	65,000
" Bessemer Bar	65,000 to	90,000

Malleable iron should always be poured from the *air furnace* and made from white charcoal iron (Nos. 3 to 5) and then annealed from ten to twenty days. This material can be used for hydraulic fittings, valves, etc., and has the tenacity given above. The *so-called* malleable iron, poured from a cupola and made of gray iron and annealed from five to ten days, has a tenacity of from 20,000 to 32,000 lbs. per square inch, and should not be used for hydraulic or machine work

Safe Loads for Bolts.

STRESS, 10,000 POUNDS PER SQUARE INCH FOR STEEL.
" 7,000 " " " " " IRON.

Diameter.	Area at Bottom of Thread.	Safe load in Steel.	Safe load in Iron.	No. of Threads per Inch.
$\frac{1}{2}$	.126	1.260	.882	13
$\frac{5}{8}$	.202	2.020	1.414	11
$\frac{3}{4}$	.30	3.000	2.100	10
$\frac{7}{8}$	.42	4.200	2.940	9
1.	.55	5.500	3.850	8
$1\frac{1}{8}$	.69	6.900	4.830	7
$1\frac{1}{4}$	.89	8.900	6.230	7
$1\frac{3}{8}$	1.06	10.600	7.420	6
$1\frac{1}{2}$	1.29	12.900	9.030	6
$1\frac{5}{8}$	1.51	15 100	10.570	$5\frac{1}{2}$
$1\frac{3}{4}$	1.75	17.500	12.150	5
$1\frac{7}{8}$	2.05	20.500	14.350	5
2	2.30	23.000	16.100	$4\frac{1}{2}$
$2\frac{1}{4}$	3.02	30.200	21.140	$4\frac{1}{2}$
$2\frac{1}{2}$	3.72	37.200	26.040	4

The Size of Bolt Heads and Nuts.

Diameter of bolt = 1.

Diameter of head and nut, square or hexagon = $1\frac{3}{4}$ from side to side.

Diameter of head and nut, hexagon = 2, over angles.

Thickness of head = $\frac{3}{4}$ of diameter of bolt.

Thickness of nut = diameter of bolt.

HYDRAULIC MEMORANDA.

A cubic foot of water contains 7.480519 gallons.

A gallon of water contains 231 cubic inches, and weighs 8.33111 pounds. (U. S. Standard.)

The friction of water in pipes is as the square of the velocity.

The height of a column of fresh water, equal to a pressure of one pound per square inch, is 2.31 feet. (In usual computation this is taken at 2 feet, thus allowing for ordinary friction.)

The mean pressure of the atmosphere is estimated at 14.7 pounds per square inch, so that with a perfect vacuum it will sustain a column of water 33.9 feet high.

To find the pressure in pounds, per square inch. of a column of water, multiply the height of the column in feet by .434 — Approximately we say that every foot elevation is equal to one-half pound pressure per square inch; this allows for ordinary friction.

In designing hydraulic machinery, a rule should be made that in all cases a leather shall work against brass or hydraulic bronze, and never against cast iron. Hemp packing should always be used to work against iron, and not come in contact with brass. Hemp packing and stuffing boxes can readily be used, with a working pressure of 1500 lbs. per sq. in.; and by the use of Du Val metallic packing in a stuffing box, with pressures as high as 5,000 lbs. per sq. in.

Working and Test Pressures.

Working Pressure.	*Test Pressure.*
400 lbs. per sq. in	1,250 lbs. per sq. in.
750 " " "	2,000 " " "
1,500 " " "	3,500 " " "
3,000 " " "	5,000 " " "
5,000 " " "	7,500 " " "

In good practice it is always well to test the parts of all hydraulic machines, etc., working at a given pressure, to the corresponding test pressure given in the table above.

Table of Gallons.

	Cubic Inches in a Gallon.	Weight of a Gallon in Pounds Avoirdupois.	Gallons in a Cubic Foot.	Weight of a cubic foot of water, English standard, 62 . 3210286 lbs. Avoirdupois.
United States	231.	8.33111	7.480519	
New York	221.81918	8.00	7.901285	
Imperial	277.274	10.00	6.232102	

Useful Numbers.

Cubic Inches	X	by	.00058	=	Cubic Feet
Circular Inches	X	"	.00546	=	Square Feet
Cubic Feet	X	"	7.48	=	U. S. Gallons
Cubic Inches	X	"	.004329	=	U. S. Gallons
Cylindrical Inches	X	"	.0034	=	U. S. Gallons
U. S. Gallons	X	"	.13367	=	Cubic Feet
Cylin. Ft. of Water	X	"	6.	=	U. S. Gallons
Cubic Ft. of Water	X	"	62.5	=	Pounds Avoirdu-
Cu. In. of Water	X	"	.03617	=	" " [pois
Cylin. In. of Water	X	"	.02842	=	" "
268.8 U. S. Gallons of water				=	One Ton
35.88 Cu. Ft. of Water				=	One Ton

Thickness of Hydraulic Cylinders.—(Formulae.)

Merriman gives the following:

s = allowable maximum stress in metal.
p = pressure in same units.
R = outside radius.
r = interior radius.
t = thickness.

$$t = \frac{r\,p}{s - p}$$

Rankine gives:

$$R = \sqrt{\frac{s + p}{s - p}} \times r$$

The foregoing formulæ apply only to "thick cylinders," and must be used with discretion, as will be seen from the particulars given below.

To bring these formulæ within practical limits at all, the allowable maximum stress, "s," should not be taken at more than 4,000 lbs. per □″ for the comparatively low values of "p," which gives "t" a value that will not make a "thick" cylinder.

From a table used by Sir W. G. Armstrong, the following thicknesses for cast-iron cylinders for a working pressure of 1,000 lbs. per □″ are taken, and these can be relied on for practical work:

Diam. of Cylinder, Inches,	2	3	4	5	6	7
Thickness of " "	0.832,	1.042,	1.146,	1.354,	1.552,	1.77,
Diam. of Cylinder, Inches,	8	9	10	11	12	13
Thickness of " "	1.875,	1.979,	2.02,	2.34,	2.578,	2.734,
Diam. of Cylinder, Inches,	14	15	16	17	18	19
Thickness of " "	2.89,	3.046,	3.19,	3.32,	3.45,	3.58,
Diam. of Cylinder, Inches,	20	21	22	23	24	
Thickness of " "	3.697,	3.802,	3.906,	4.01,	4.114	

For any other pressures, multiply by the ratio of that pressure to 1,000.

Mr. Wm. Kent says: "These figures correspond nearly to the formula $t = 0.175\ d + 0.48$, in which t = thickness, and d = diameter in inches up to 16″ diameter, but for 20 inches diameter, the addition of 0.48 is reduced to 0.19, and at 24″ it disappears."

Cast-iron should not be used for pressures exceeding 2,000 lbs. per □″, and it is better to use steel castings or forged steel for cylinders which would be over 6″ thick in cast iron.

Capacities of Cylinders and Rams.

Diameter. Inches.	Area.	Load at 1,500 lbs. □" in lbs.	Cubic Ins. per Foot of Cylinder	Gallons per Foot of Cylinder.	Gallons per Inch of Cylinder.
1	.7854	1,178	9.42	.040	.0033
1⅛	.9940	1.491	11.93	.051	.0042
1¼	1.227	1,840	14.72	.063	.0052
1⅜	1.484	2,227	17.8	.077	.0064
1½	1.767	2,650	21.20	.091	.0076
1⅝	2.073	3,110	24.87	.107	.0089
1¾	2.405	3,607	28.86	.124	.0103
1⅞	2.761	4,140	33.13	.143	.0119
2	3.141	4,711	37.69	.163	.0136
2⅛	3.546	5,319	42.55	.184	.0153
2¼	3.976	5,964	47.71	.206	.0171
2⅜	4.430	6,645	53.16	.230	.0191
2½	4.908	7,362	58.90	.254	.0211
2⅝	5.411	8,116	64.93	.281	.0234
2¾	5.939	8,908	71.26	.308	.0257
2⅞	6.491	9.736	77.89	.337	.0281
3	7.068	10,602	84.81	.367	.0306
3¼	8.295	12.442	99.54	.430	.0358
3½	9.621	14,431	115.45	.50	.0416
3¾	11.04	16,560	132.48	.573	.0477
4	12.56	18,840	150.72	.652	.0543
4¼	14.18	21,270	169.92	.735	.0612
4½	15.90	23.850	190.80	.826	.0688
4¾	17.72	26.580	212.64	.920	.0767
5	19.63	29,445	235.56	1.02	.0850
5¼	21.64	32,460	259.68	1.124	.0936
5½	23.75	35,625	285.0	1.23	.1025
5¾	25.96	38,940	311.52	1.348	.1123
6	38.27	42,405	339.24	1.468	.122
6¼	30.67	46,005	368.04	1.59	.132
6½	33.18	49.770	398.16	1.72	.143
6¾	35.78	53.670	429.36	1.85	.154
7	38.48	57,720	461.76	1.99	.165
7¼	41.28	61.920	495.36	2.14	.178
7½	44.17	66,255	530.04	2.29	.190

Capacities of Cylinders and Rams.—*Continued.*

Diameter. Inches.	Area	Load at 1,500 lbs □″ in lbs.	Cubic Ins per Foot of Cylinder.	Gallons per Foot of Cylinder.	Gallons per Inch of Cylinder.
7¾	47.17	70,755	566.04	2.45	.20
8	50.26	75,390	603.12	2.61	.217
8¼	53.45	80,175	641.40	2.77	.230
8½	56.74	85,110	680.88	2 94	.245
8¾	60.13	90,195	721.56	3.12	.260
9	63.61	95,415	763.32	3.30	.275
9¼	67.20	100,800	806.40	3.49	.290
9½	70 88	106,320	850.56	3.68	.306
9¾	74.66	111,990	895.92	3.87	.322
10	78.54	117,810	942.48	4.08	.340
10½	86.59	129,885	1039.0	4.50	.375
11	95.03	142,545	1140.3	4.93	.410
11½	103.8	155,700	1245.6	5.39	.449
12	113.0	169,500	1356.0	5.87	.489
12½	122.7	184,050	1472.4	6.37	.530
13	132.7	190,050	1592.4	6.89	.574
13½	143.1	214,650	1717.2	7.43	.619
14	153.9	230,850	1846.8	8.00	.666
14½	165.1	247,650	1981.2	8.57	.714
15	176.7	265,050	2120.4	9.17	.764
15½	188.6	282,900	2263.2	9.80	.816
16	201.0	301,500	2412.0	10.44	.87
16½	213.8	320,700	2565.6	11.10	.925
17	226.9	340,350	2722.8	11.78	.981
17½	240.5	360,750	2886.0	12.49	1.04
18	254.4	381,600	3052.8	13.21	1.10
18½	268.8	403,200	3225.6	13.96	1.16
19	283.5	425,250	3402.0	14.72	1.22
19½	298.6	447,900	3583.2	15.51	1.29
20	314.1	471,150	3769.2	16.31	1.36

HYDRAULIC PRESSURE TRANSMISSION.

INTRODUCTION.

Water under high pressure (700 lbs. per □″ and upward.) affords a ready and satisfactory method of transmitting power to a distance, and is particularly adapted to the movement of heavy loads at moderate velocities, by cranes and elevators. It is also the most efficient way of operating large tools for pressing and joining metals, as in riveting, flanging, punching, shearing, forging and also for the operation of turrets in battle-ships, and of disappearing gun carriages, high duty elevator plants, etc., etc.

That it can be made to operate tools quickly is shown from the fact that forging presses are now made to make from 50 to 60 strokes per minute. The system usually consists of one or more pumps, capable of developing the required pressure; accumulators, which are vertical cylinders with heavily weighted plungers passing through stuffing boxes, by which a quantity of water may be accumulated at the pressure to which the plunger is weighted; and of the distributing mains to the presses, cranes or other machinery to be operated.

MECHANICAL VALUE OF WATER UNDER ACCUMULATOR PRESSURE.

The gross amount of energy of the water under pressure (stored in the accumulator.) measured in foot pounds, is its volume in cubic feet × its pressure in pounds per square foot. The horse power of a given quantity, steadily flowing, is

$$\text{H. P.} = \frac{144 \quad p\ Q}{550} = .2618\ p\ Q$$

in which Q = quanity flowing in feet per second,
and p = pressure in pounds per square inch.

Theoretically, the mechanical value of water under an accumulator pressure of 700 lbs. per □″ (549.78 per circular inch), is 100,800 foot lbs., or 45 foot tons per cubic foot of water, irrespective of the time in which it is consumed. This gives 3.0545 H. P. per cu. ft. per minute, or one H. P. requires 0.32738 per cu. ft per minute. Approximately, this equals 1 H. P. from 2 gallons of water, but practically, allowing for all losses, 3½ gallons are required.

THE EFFICIENCY OF HYDRAULIC APPARATUS.

The useful effect of a direct-acting cylinder, ram or plunger is usually taken at 93 per cent., though in practice it is found that the friction loss varies from 5 per cent. to 18 per cent. according to the condition of the packing, which makes the Armstrong practice of 86 per cent. safer to use in ordinary work. The following table is given by Mr. Percy Westmacott as the efficiency of a ram with chain and pulley multiplying gear, properly proportioned and well lubricated.

Multiplying	2 to 1	4 to 1	6 to 1	8 to 1	10 to 1	12 to 1	14 to 1	16 to 1
Efficiency, per cent.	80	72	72	67	63	59	54	50

With large sheaves, small steel pins and wire rope, the efficiency has been found as high as 66 per cent. for a multiplying power of 20 to 1.

Henry Adams gives the following formula for effective pressures in cranes and hoists; it will be found to correspond with the above.

P = Accumulator pressure in lbs. per □″
m = ratio of multiplying power.
E = effective power in lbs. per □″, including all allowances for friction.

$$E = P\ (.84 - .02\ m)$$

SPEED OF HOISTING BY HYDRAULIC POWER.

The maximum allowable speed for warehouse cranes is 6 ft. per second; for platform cranes and lifts, 4 ft. per second; for passenger and wagon hoists with heavy loads, 2 feet per second. Mr. Henry Adams is authority for the statement that "The maximum speed under any circumstances should never exceed 600 ft. per minute." But recent elevator practice has shown that a speed of from 700 to 800 ft. per minute can be attained.

VELOCITY OF WATER THROUGH PIPES AND VALVES.

The following table gives the velocity in feet per second at pressures from 700 lbs. to 5,000 lbs., which the water will acquire if discharged into the atmosphere through an approximately frictionless aperture. Velocities for intermediate pressures may be calculated from the formula:—

Velocity in feet per second = $12.19 \sqrt{\text{pressure in lbs. □″}}$.

In lbs. Pressure per Square Inch.	700	1500	2000	3000	4000	5000
Velocity in feet per second............	326	472	548	670	772	862

With an accumulator pressure of 700 lbs. per □″ the natural velocity (theoretically) is 326 ft. per second. It is found in

practice that not more than one-tenth of this can be obtained through the pipes and one third through the valves, in order to maintain the proper speed of the machinery. The loss from friction in the pipes is about 1 lb. per □″ per 100 feet in length, after they have been laid some time, 1 lb. additional for each bend, and 10 lbs. for each branch. It is usual in calculating to take the speed of the water through valves at not more than 98 feet per second.

Experiments with water at 1600 lbs. per □″ flowing into a flanging press cylinder 20″ diameter, through a ½″ pipe contracted at one point to ¼″, gave a velocity of 114 feet per second in the pipe, and 456 feet at the reduced section. Through a ½″ pipe reduced to ⅜″ at one point, the velocity was 213 feet per second in the pipe, and 381 at the reduced section. In a ½ inch pipe without contraction, the velocity was 355 feet per second.

AREAS OF VALVES FOR MACHINERY UNDER ACCUMULATOR PRESSURE.—(FORMULAE.)

A = Area of lifting ram.
m = Ratio of multiplying power.
v = Velocity of load in feet per second.
V = " " water through valve in feet per second.
W = Length of ram, cross-head. sheaves, chain, etc., in lbs.
a = Area of lifting valve (mitred spindle).
a′ = " " lowering " " "

$$a = \frac{A\ v}{m\ V} \qquad a' = \frac{A\ v}{m\sqrt{13.8\ \frac{W}{A}}}$$

When cylinder is horizontal, then

$$\frac{W}{700} = \text{area of returning ram.}$$

AREA OF PORTS IN SLIDE VALVES.—(FORMULAE.)

(V-SHAPED OPENINGS.)

v = velocity of load in feet per second.
m = Multiplying power.
A = Area of ram in □″

$$\text{Area of pressure port} = \frac{A\ v}{98\ m}$$

$$\text{Area of exhaust port} = \frac{1.5\ A\ v}{98\ m}$$

LIFTING RAMS FOR HYDRAULIC CRANES.—(FORMULAE.)

W = Load to be lifted in pounds.
w = Weight of ram, cross-head, chain or sheaves.
l = Height of lift in feet.
m = Multiplying power.
e = Coefficient of effect = (.84 — .02 m)
a = Area of ram in □″
p = Accumulator pressure in lbs. per □″

For horizontal cylinders: $a = \dfrac{W\ m}{p\ c}$

" vertical " $a = \dfrac{W\ m + w}{P\ c}$

" Inverted " $a = \dfrac{W\ m - w}{p\ c}$

TURNING RAMS FOR HYDRAULIC CRANES.—

(FORMULAE.)

w = load in tons.
k = rake in foot
l = length between bearings in feet.
d = diameter turning drum in feet.
p = accumulator pressure in lbs. per □″
m = multiplying power. (usually 2 to 1.)
a = area turning ram per □″
f = a constant.

$$a = \frac{W\ R\ 2\ f\ m}{l\ d\ p}$$

f = 70 for cranes up to $1\frac{1}{2}$ tons
f = 60 " " " " 4 "
f = 50 " " " " 10 "

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